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# Crack Detection for Aerospace Quality Spur Gears

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# Crack Detection for Aerospace Quality Spur Gears

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## ABSTRACT

Health and Usage Monitoring System research and development involves analysis of the vibration signals produced by a gearbox throughout its life. There are two major advantages of knowing the actual lifetime of a gearbox component: safety and cost. Three spur gears were machined with a notch to provide a seeded fault. These gears were then run until tooth failure while recording the vibration signals. Standard vibration diagnostic parameters are calculated and are presented. The results of this study indicate that the detection methods examined are not robust or repeatable. Current techniques show that the cracks progressed at a much faster rate than anticipated which reduced available time for detection.

## INTRODUCTION

There is considerable work being performed in Health and Usage Monitoring Systems (HUMS) to reduce maintenance of mechanical components such as gearboxes and to increase vehicle safety. Health and Usage Monitoring can be classified into two major areas: diagnostics and prognostics. Diagnostics deals with the consistent and accurate detection of damage, while prognostics includes both damage estimation and the estimating the remaining useful life.

Diagnostics can be based on various types of data, including vibration, acoustic emission, and oil debris analysis. A large portion of the vibration diagnostics work is currently based on techniques such as fuzzy logic, neural networks, and data fusion, to name a few. Diagnostics techniques can be classified into feature extraction and detection. Feature extraction is the separation of the desired features of interest from extraneous information. The process of interpreting the remaining data is known as feature detection.

An accurate health and usage management system (HUMS) would warn of impending failure as well as provide maintenance information to appropriate support personnel. Typically, components are removed based on a conservative statistical life usually measured in hours of operation. There is no distinction of whether those hours are at ground idle or at full power. If, by monitoring the loading characteristics, an individual rotorcraft is known to be lightly loaded, it should be possible to extend the overhaul life of the transmission. This would allow more of the available life of the components to be safely used. This would require significant cooperation between the HUMS developer, the airframe manufacturer and the aircraft certification authority (i.e., U.S. Federal Aviation Agency, United Kingdom Civilian Aviation Agency, etc.). If the loading history indicates frequent heavy loading, a HUMS system would reduce the probability of an accident. Notice of an impeding failure would allow the repair of the drive system before an accident. In addition, the HUMS unit could be programmed to warn the maintenance team, several hours in advance, that specific maintenance will be required. This would allow for better scheduling of resources, thereby saving money. [1]

A major concern of current HUMS systems is their reliability. A recent report proposes that the current fault detection rate of a vibration-based system is 60 percent. A false alarm is typically generated every hundred hours. [2,3]

Since 1988, the NASA Glenn Research Center has been working on improving gear damage detection using vibration monitoring. Most of the effort has focused on pitting and other surface distress failures. Later, the testing expanded into both oil debris monitoring-based HUMS as well as vibration based crack detection and propagation. Gear cracks, although potentially more catastrophic, are much less common, thus more difficult to study.

The study of vibration diagnostics was initiated in the late 1970s. There was research performed in both the

United States and in the United Kingdom. The approaches used were fundamentally different.

The United States Department of Defense sponsored research in the specific area of helicopter gearboxes. These techniques were, for the most part, based on the precise analog filtering of the time domain signal. The analog systems had to be tuned to particular frequencies and could not be adjusted for variation in speed or torque. This required gearboxes that were relatively simple, containing one gear mesh with only a slight speed reduction. Most often these were tail rotor drive shaft gearboxes.

To allow a system to be more responsive to variations in speed and torque, the researchers in the United Kingdom focused on digital based analysis. At this time, the personal (or mini) computer was beginning to emerge. This, combined with the availability of PC-based analog to digital converter boards, allowed the analysis to shift into the digital domain. One of the earliest successful attempts at spectral analysis was performed in the United Kingdom. For the most part, current research is the evolution of the concepts pioneered in the U.K. [3] The frequency domain signal processing techniques are described in detail in the next section.

Many different techniques have been proposed to detect damage in mechanical power transmissions. These methods include vibration, oil debris detection, chemical element detection, and acoustic emission. The focus of this paper is the analysis of the vibration. There are two major requirements of any HUMS unit. First, it must identify that there is, in fact, a fault in the subsystem. For the most part, it is not critical to determine the type of fault and which component is faulty (although for maintenance purposes, this is highly desirable). Secondly, detection accuracy is critical. The HUMS warning will be ignored if too many false alarms are generated.

The general procedure for health monitoring using vibration signals in a steady state system is relatively simple. There are five distinct elements: 1) signal acquisition, 2) synchronous averaging, 3) feature detection and extraction, 4) interpretation of results, and 5) prognosis. The first two, signal acquisition and synchronous averaging are relatively straightforward. The greatest amount of work to this point has been in the areas of feature extraction and detection as well as interpretation of the results. Prognosis deals with the prediction of how much useful life remains in a damaged component. An accurate prognosis would prevent stranding crew and passengers in a potentially hazardous location.

## THEORY OF GEAR FAILURE DETECTION METHODS

The traditional methods of gear failure detection methods are typically based on some statistical measurement of vibration energy. The primary differences are based on which of the characteristic frequencies are included, excluded, or used as a reference. [4]

### Root Mean Square

The root mean square (RMS) is a simple measure of the effect of a fluctuating signal (Eq. (1)). It was originally developed to characterize the heating of a resistor subjected to a sine wave alternating current. RMS is defined to be the square root of the average of the sum of the squares of an infinite number of samples of the signal. It is also sometimes referred to as the standard deviation of the signal average. For a simple sine wave, the RMS value will be defined to be approximately 0.707 times the amplitude of the signal.

$$\mathrm{RMS} = \sqrt{\frac{1}{N}\left[\sum_{i=1}^{N}(S_i)^2\right]} \tag{1}$$

### Crest Factor

The Crest Factor (CF), shown in Eq. (2), is calculated by dividing the maximum positive peak value by the RMS value of the signal. [5] This makes the parameter a normalized measurement of the amplitude of the signal. A signal that has a few, high amplitude peaks would produce a greater Crest Factor as the numerator would increase (high amplitude peaks), as the denominator decreases (few peaks means lower RMS).

$$\mathrm{CF} = \frac{S_{0-pk}}{\mathrm{RMS}} \tag{2}$$

### Energy Operator

The Energy Operator [6], is a parameter that is a simple calculation. The input signal for each point in time is squared and the product of the point before and after is subtracted. In the case of the endpoints, the data is looped around. Specifically, the when calculating the first point, use the last point and vice versa. The normalized kurtosis of the resultant signal is then taken and reported as the energy operator.

### Kurtosis

The kurtosis (Eq. (3)) is simply the normalized fourth moment of the signal. [7] The moment is normalized to the square of the variance of the signal. The kurtosis is a statistical measure of the number and amplitude of peaks in a signal. That is, a signal that has more and sharper peaks will have a larger value. A Gaussian distribution has a kurtosis value of very nearly three. It turns out that a gearbox in good condition should emulate a Gaussian distribution, and therefore have a value near three. It should be noted that investigators subtract three from this calculated value. This produces a value of zero for a gearbox in good condition.

$$\text{Kurtosis} = \frac{N\sum_{i=1}^{N}(S-\bar{S})^4}{\left[\sum_{i=1}^{N}(S-\bar{S})^2\right]^2} \tag{3}$$

where

- $S$ signal
- $\bar{S}$ mean value of signal
- $I$ data point number in time record
- $N$ number of data points

**M6**

The M6 parameter [8], shown in Eq. (4), is a continuation of the kurtosis. In this particular case, it is the sixth moment that is used. It is normalized in a similar manner as the kurtosis, except that the variance now has to be raised to the third power. In general, the characteristics of the spread of the distribution show up to be even (as opposed to odd) functions of the statistical moment. The odd functions relate the position of the peak density distribution with respect to the mean.

$$M6 = \frac{N^2\sum_{i=1}^{N}(d-\bar{d})^6}{\left[\sum_{i=1}^{N}(d-\bar{d})^2\right]^3} \tag{4}$$

where

- $d$ difference signal
- $\bar{d}$ mean value of difference signal
- $i$ data point number in time record
- $N$ number of data points

**Energy Ratio**

Heavy uniform wear can be detected by the energy ratio. [5] The difference signal (d) is the resultant signal after the regular meshing components (r) (mesh and harmonic frequencies) are removed. It compares the energy contained in the difference signal to the energy contained in the regular components signal. The theory is that as wear progresses, the energy is moved from the regular signal to the difference signal. (Eq. (5))

$$ER = \frac{RMS_d}{RMS_r} \tag{5}$$

**FM0**

The Zero-Order figure of merit, FM0, shown in Eq. (6), detects significant changes in the time synchronous average. It was first proposed by Stewart [3]. It is a technique that is gives no information about where in the spectrum the damage is located. It compares the peak to peak value of the signal to the sum of the RMS values of the mesh frequency and its harmonics.

$$FM0 = \frac{S_{pk-pk}}{\sum_{i=1}^{N} RMS(f_i)} \tag{6}$$

where

- $S_{pk\text{-}pk}$ peak to peak value of signal
- $f_i$ mesh frequency and harmonics
- $N$ number of harmonics + 1

**FM4**

The FM4 vibration diagnostic parameter (Eq. (7)) is one of the most popular parameters used. [3] This parameter detects changes in the vibration resulting from damage limited to several teeth. A difference signal is created for a data record by removing the shaft and meshing frequencies, their harmonics, and the first order sidebands in the frequency domain. The kurtosis (fourth statistical moment) is calculated by dividing the kurtosis by the square of the variance of the difference signal. The FM4 parameter is non-dimensional and is calculated by dividing the kurtosis by the square of the variance of the difference signal of a gearbox in good condition and is also approximately three. As localized damage begins in a gearbox, the FM4 value increases.

$$FM4 = \frac{N\sum_{i=1}^{N}(d_i-\bar{d})^4}{\left[\sum_{i=1}^{N}(d_i-\bar{d})^2\right]^2} \tag{7}$$

where

- $d$ difference signal
- $\bar{d}$ mean value of difference signal
- $N$ total number of points in time record
- $i$ data point number in time record

**NA4**

The NA4 parameter (Eq. (8)) was developed to overcome a shortcoming of the FM4 parameter. [4] As the occurrences of damage progresses in both number and severity, FM4 becomes less sensitive to the new damage. Two changes were made to the FM4 parameter to develop the NA4 parameter as one that is more sensitive to progressing damage. One change is that FM4 is calculated from the difference signal while NA4 is calculated from the residual signal. The residual signal includes the first order sidebands that were removed from the difference signal. The second change is that trending was incorporated into the NA4 parameter. While FM4 is calculated as the ratio of the kurtosis of the data record divided by the square of the variance of the same data record, NA4 is calculated as the ratio of the kurtosis of the data record divided by the square of the average variance. The average variance is the mean value of the variance of all previous data records in the run ensemble. These two changes make the NA4 parameter a more sensitive and robust parameter. The NA4 parameter is calculated by

$$NA4 = \frac{N\sum (r_i - \bar{r})^4}{\frac{1}{M}\sum_j \left[\sum_i (r_{ij} - \bar{r}_j)^2\right]^2} \tag{8}$$

where

- $r$ residual signal
- $\bar{r}$ mean value of residual signal
- $N$ total number of points in time record
- $M$ current time record in run ensemble
- $i$ data point number in time record
- $j$ time record number in run ensemble

**NB4**

The NB4 parameter is the time-averaged kurtosis of the envelope of the signal that is bandpass filtered about the mesh frequency. [9] An estimate of the amplitude modulation caused by the sidebands of the meshing frequency, is calculated using the Hilbert Transform. The Hilbert transform creates a complex time signal in which the real part is the bandpassed signal and the imaginary part is the Hilbert transform of the signal.

**NA4***

As damage progresses from localized to distributed, the variance of the kurtosis increases dramatically. Since the kurtosis is normalized by the variance, this results in the kurtosis decreasing to normal values even with damage present. To counter this effect, NA4* was developed. [10] While the kurtosis for a data record is normalized by the squared average variance for the run ensemble for NA4, with NA4* the kurtosis for a data record is normalized by the squared variance for a gearbox in good condition. This is a change in the trending of the data and was proposed to make a parameter that is more robust as damage progresses.

In order to estimate the variance for a gearbox in good condition, a minimum number of data records of a run ensemble is chosen to ensure a statistically significant sample size. The variance of the residual signal for all data records is calculated, as well as the mean and standard deviation. The mean is used as the current estimate of the variance for a gearbox in good condition. When the next data record is available, a judgment is made as to whether to include that data record as representative of a good gearbox. A gearbox with damaged gears will have a larger variance that one in good condition. The decision is based on an upper limit L (Eq. (9)), which in turn is dependent on the choice of a probability coefficient Z, and is calculated by

$$L = \bar{x} + \frac{Z}{\sqrt{n}}\sigma \tag{9}$$

where

- $\bar{x}$ mean value of previous variances
- $Z$ value for a normal distribution
- $\sigma$ standard deviation of previous variances
- $n$ number of samples ($n \geq 30$)

The value for the Z parameter can be found in introductory statistics books. If the current variance exceeds this limit, then it is judged that the gearbox is no longer in "good" condition and the previous estimate of the variance is used for the remainder of the run ensemble. If the variance for the new data record does not exceed this limit, then the new data record is included into the data representing the gearbox in good condition.

The decision of what probability coefficient is chosen is based on many factors. The most difficult trade-off is that of Type I or Type II errors. A Type I errors is an undetected defect. A Type II error, on the other hand, reports damage when none is present. The choice of the probability coefficient is a compromise between having too many Type II errors and not detecting damage.

**NB4***

The diagnostic parameter NB4* parameter is the addition of the run ensemble averaging and the statistical limitation of the growth of the square of the variance first introduced in the development of NA4*. The calculation of the numerator of this parameter remains the same as in NB4. The denominator does have the averaging effect of NA4*, and determines if the current variance is of sufficient probability to be contained in the previous samples.

### FM4*

The diagnostic parameter FM4*parameter is, like NB4*, the addition of the run ensemble averaging and the statistical limitation of the growth of the square of the variance. The calculation of the numerator of this parameter remains the same as in FM4. The denominator has the averaging effect of NA4*, and also determines if the current variance is of sufficient probability to be contained in the previous samples.

## EXPERIMENTAL CONFIGURATION

### Facility Description

A spur gear fatigue test stand at the NASA Glenn Research Center in Cleveland, Ohio was used to perform the testing. This facility, shown in Figure 1, allows the study of effects of gear tooth design, gear materials and lubrication on the fatigue lives of aerospace quality gears. The test stand operates using the closed loop torque regeneration principle. The test gears are connected by shafts to a pair of helical gears that complete the loop. The torque is applied through a hydraulic loading mechanism that twists one slave gear relative to the shaft that supports it. Therefore the torque is usually reported as a function of the hydraulic pressure. The drive motor only has to supply enough power to overcome the losses in the system. The test gears are lubricated with an independent oil system. The speed, torque, and input oil test temperatures can all be controlled.

During health monitoring tests, an infrared optical sensor monitors the input shaft using a timing mark. Typically, there are two accelerometers used for HUMS research, one mounted on the outside of the test housing, with the other mounted in the test section directly on the bearing cover plate.

### Test gear description

The spur gear test rig uses a pair of spur gears having 28 teeth, a pitch diameter of 88.9 mm (3.50 inch), and a face width of 6.35 mm (0.25 inch). During a surface fatigue test, the gear faces are usually offset by 2.79 mm (0.11 inch) to allow a higher surface stress without a corresponding increase in the bending stress. For these tests however, the gears were in contact across the full face width. The tests were also run at a higher torque than normal. A photograph of a crack test gear is shown in Figure 2.

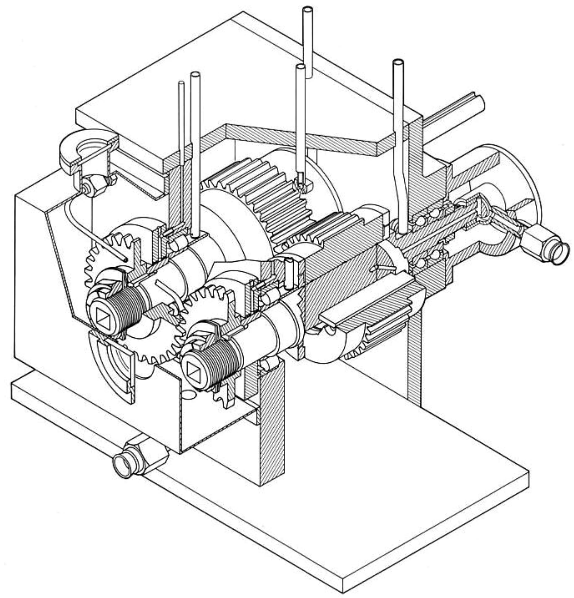

Figure 1. Spur Gear Test Apparatus

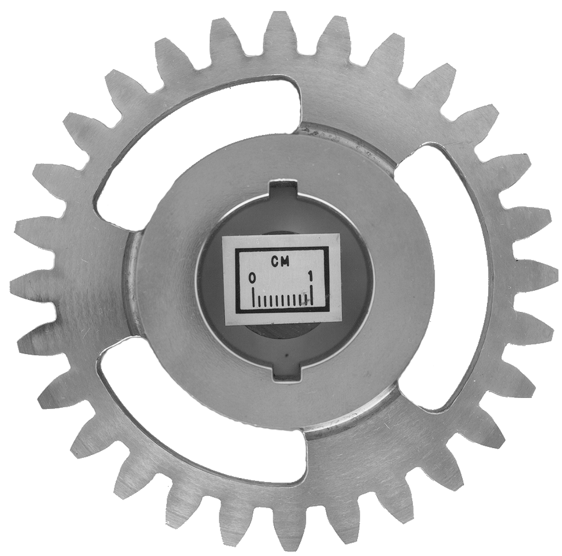


Figure 2. Representative gear for crack tests

### Notch geometry

A notch was machined in the root area of the gear to provide a concentrated flaw from which a crack could initiate. This location was chosen since this is the point of highest tensile bending stress on the gear tooth surface. The higher stress provides the best opportunity for crack propagation.

The notch traversed the entire face width of the gear and was created using electrical discharge machining (EDM) process. (Figure 3) This process was chosen for its ability to control the size of the notch. The size of the notch is controlled by both the shape and electrical current of the electrode and is typically 0.254 mm (0.010 inch) deep.

**Accelerometers**

Two research accelerometers were mounted on the test gearbox. The first one, (and only one for the first test) was located on the housing of the gearbox. The location was chosen based upon previous modal analysis testing on an identical gearbox. [11] In this paper, this accelerometer is noted as the "case" accelerometer. It is piezoelectric with a frequency response from 20 Hz to 50 kHz. The second accelerometer is also piezoelectric, but smaller and has a frequency range from 1 Hz to 10 kHz. This is mounted 30 degrees clockwise from the vertical centerline for the right (driven) shaft on the bearing retention cap inside the gearbox. The location is in the load zone of the bearing and provides the most direct transfer path for the vibration to travel. This accelerometer is referred to as the "shaft" accelerometer. The configuration is shown in Figure 4.

**Tachometer**

The once per revolution tachometer signal is generated using an infrared optical sensor that is located on the input shaft to the test gearbox. The sensor detects a change in the reflectivity of an infrared light. The connecting shaft has a piece of highly reflective silver colored tape cemented to the black oxide coated shaft. This provides a reliable signal that has good dynamic performance.

## RESULTS

These tests were run at an overloaded condition to accelerate testing. It will be shown that it is difficult to determine crack initiation on these gears. It would be beneficial to run the tests at overloaded conditions to initiate a crack, and then reduce the load to observe stable crack growth. This would allow a more accurate study of the vibration signature during the critical crack growth period.

During the first test , only one accelerometer was used. This was the case mounted accelerometer. The "shaft" accelerometer was installed between the first and second tests, and was available for the remainder of the tests.

**Test 1**

This test, run at 124.7-154.6 Nm (92-114 ft-lb) and 2500 rpm produced a tooth fracture (Figure 5) after almost 237 hours. The original notch is readily visible in the fillet on the left side of the gear tooth. The crack initiated at the edge of the notch and progressed to the fillet on the right. Thirteen of the most often used diagnostic parameters are shown in using the data from the "case" accelerometer. None of these parameters detected the tooth fracture.

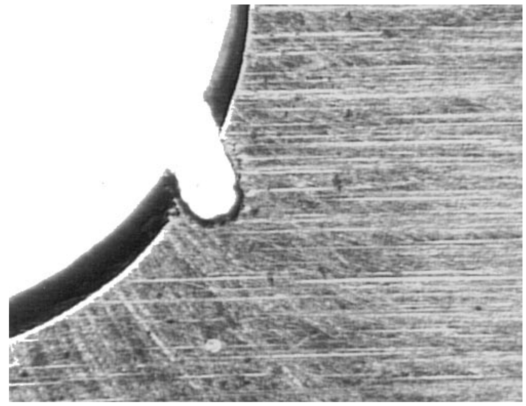

Figure 3. Notch in gear tooth

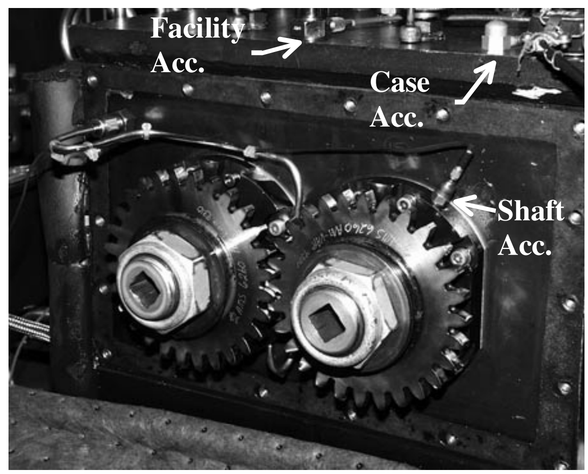


Figure 4. Gearbox accelerometer mounting locations

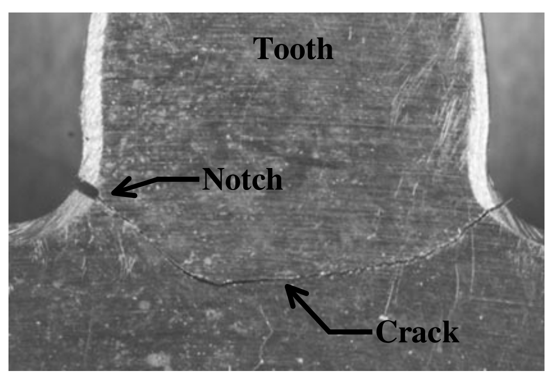


Figure 5. Gear tooth fracture after test 1

Figure 6 also shows the results of when the facility accelerometer lost power and shut the facility down (approximately 70 hours), and an unexplained set of conditions at about 170 hours. Experience has shown that several of the diagnostic parameters take a significant amount of time to settle back into steady state like conditions after an interruption, if at all. It is important to note the amplitude of these disturbances for comparison later on. It is proposed that during a shutdown, the temperature decrease changes the system dynamics by altering the clearances and contact stresses from the previous conditions. In this figure there is no obvious indication of crack initiation, progression or separation of the gear tooth.

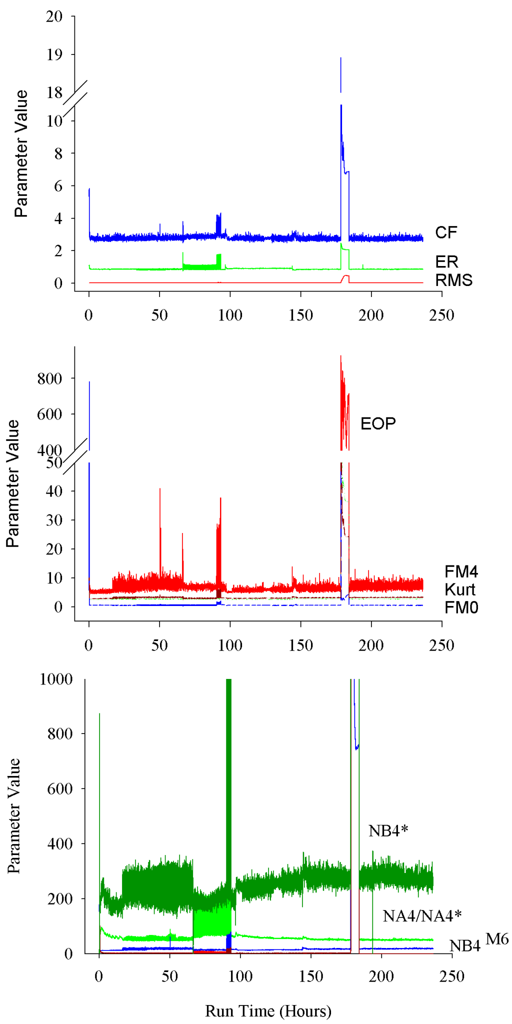


Figure 6. Test 1, case accelerometer parameters

**Test 2**

Test 2 was conducted at 5000 rpm and 154.6 Nm (114 ft-lbs) torque. This test ended at 1.7 hours with a fracture through the rim (Figure 7), which may have been caused by running near a gear resonance. At 1.4 hours, high vibration levels caused a test shutdown. The gear was examined and a mark taken to be dirt or fuzz was noticed. This may have actually been the crack that eventually propagated through the rim. The notch can be seen in the upper left corner of Figure 7.

Figures 8 and 9 present the results of applying these parameters to the vibration recorded by the two accelerometers. In this test, almost all of the techniques examined indicate something at 1.25 hours. The variations due to the shutdown and subsequent startup are readily visible.

The ideal parameter would show a step change at initiation of damage, a linear increase during damage progression with another step increase to a high level to indicate the loss of the tooth for the remainder of the run. The M6 parameter demonstrates one of the deficiencies of several of the parameters. Some of the parameters after increasing to indicate damage, reduce in value as the damage becomes more distributed. If the peak is not detected, there is a very real possibility of encountering a Type I error. The results of the NB4* parameter, at least in this test, demonstrates the desired characteristics of a robust parameter.

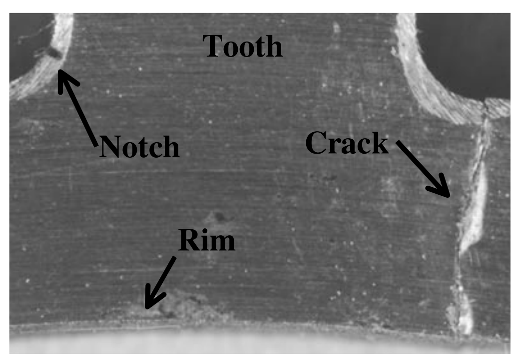


Figure 7. Gear rim fracture after test 2

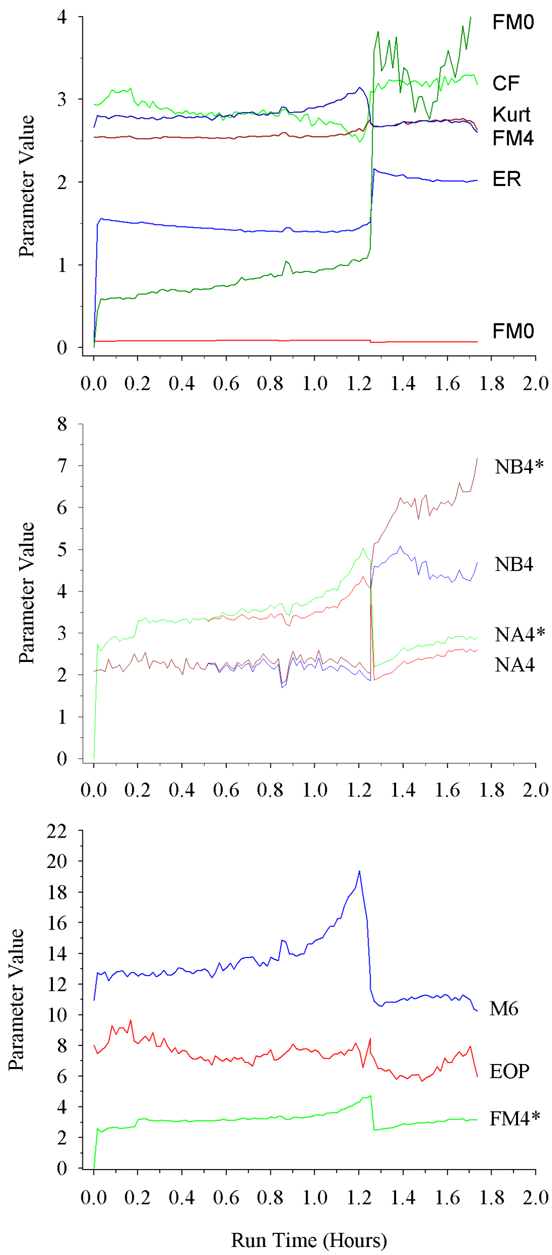


Figure 8. Test 2, case accelerometer parameters

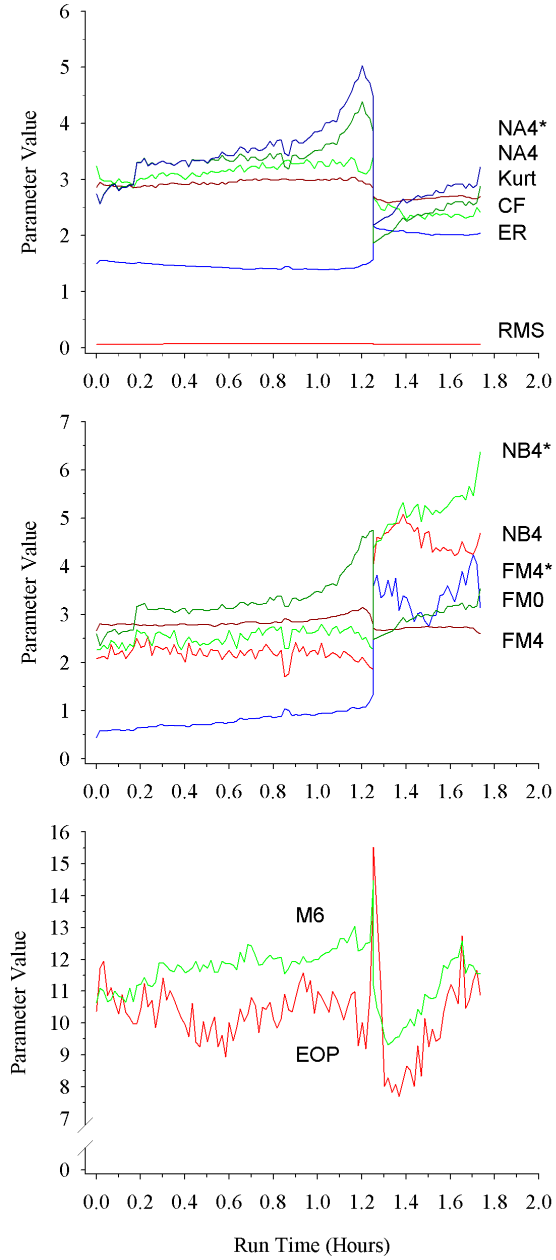


Figure 9. Test 2, shaft accelerometer parameters

## Test 3

This test also produced a fractured tooth (Figure 10). This fracture was not complete and progressed about two-thirds of the width of the tooth. The facility monitoring accelerometer detected a high vibration level due to the crack and shut down the system before the loss of the tooth. The shutdown occurred after almost 420 hours of 4925 rpm at torques of 124.7, 139.4, and 154.6 Nm (92, 106, and 114 ft-lbs) of torque. The gear was then later run at various torques until complete fracture occurred.

As seen in Figure 11 and Figure 12, the widely used parameters do not readily indicate any crack initiation or propagation. The last 3 points of FM0 in Figure 11 and NA4 and FM0 in Figure 12, with their abrupt increase in value, may hint at the damage. In this case, the shaft mounted accelerometer, with its shorter and more direct transfer path indicates the damage better

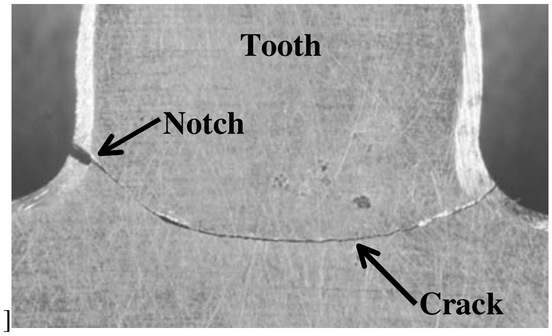


Figure 10. Gear tooth fracture after test 3

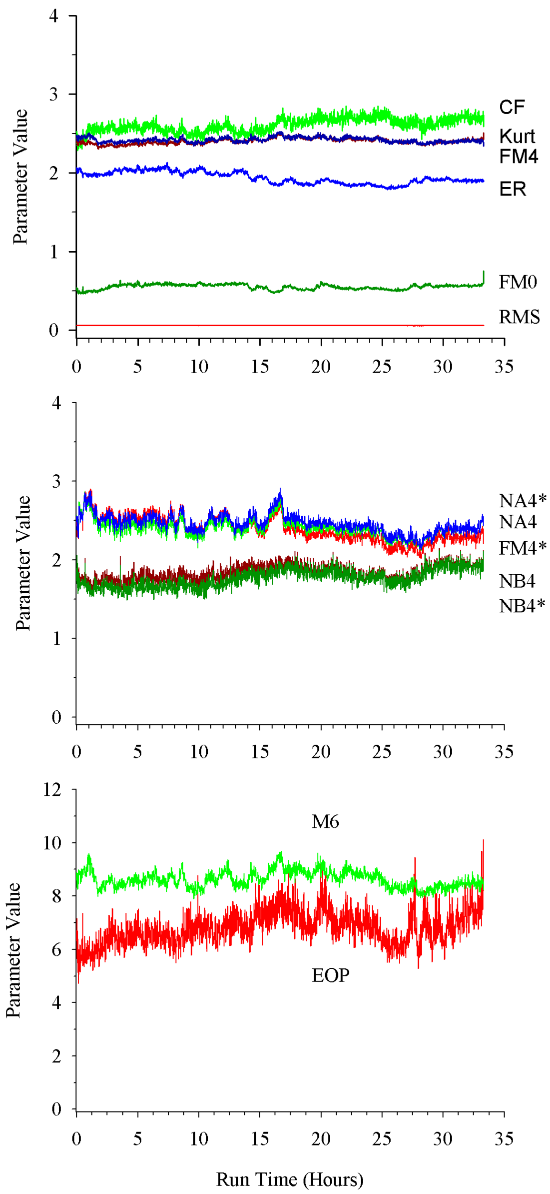


Figure 11. Test 3, case accelerometer parameters

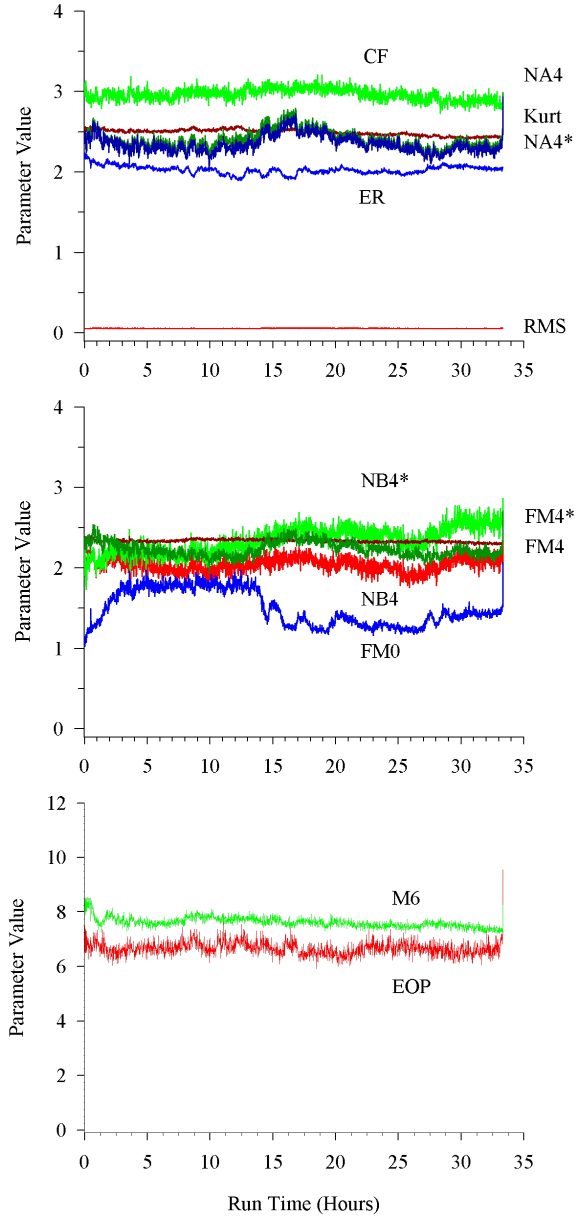


Figure 12. Test 3, shaft accelerometer parameters

## CONCLUSIONS

The tests conducted in this study reflect other previous experiments that show that no individual technique routinely outperforms the others for gear crack detection. Several methods for feature extraction and detection appear to be required. At times, some failures are not detected. This leads to several important conclusions that can be obtained from this testing:

1. For the commonly used vibration diagnostic parameters examined here, there is no single parameter that will reliably and accurately detect gear fractures until there is significant, possibly secondary damage (complete loss of tooth).

2. The techniques presented in this paper, while improving on existing techniques, still do not have sufficient robustness and accuracy. They may, however, provide the feature extraction necessary for future detection algorithms.

3. Current techniques sometimes respond better to speed, torque, and other changes in the dynamic system than the changes in the condition of the gears. Temperature fluctuations (and the resultant changes in the dynamics of the system) that occur when the gearbox is shut down may cause false indications of damage that mask the effects of the gear damage.

4. Using current techniques, it is almost impossible to be able to reliably detect a tooth fracture in sufficient time to be able to monitor its growth.

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Health and Usage Monitoring System research and development involves analysis of the vibration signals produced by a gearbox throughout its life. There are two major advantages of knowing the actual lifetime of a gearbox component: safety and cost. In this report, a technique is proposed to help extract the critical data and present it in a manner that can be easy to understand. The key feature of the technique is to make it independent of speed, torque and prior history for localized, single tooth damage such as gear cracks. This extraction technique is demonstrated on two sets of digitized vibration data from cracked spur gears. Standard vibration diagnostic parameters are calculated and presented for comparison. Several new detection algorithms are also presented. The results of this study indicate that crack detection methods examined are not robust or repeatable. The proposed techniques provide a limited improvement to existing diagnostic parameters. Current techniques show that the cracks progressed at a much faster rate than anticipated which reduced available time for detection.

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